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THE CHARACTERISTICS OF SMALL CAPACITY SCROLL COMPRESSORS FOR RESIDENTIAL AIR CONDITIONERS

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ABSTRACT

The sealing effect of lubricating oil against gas leakage has been experimentally investigated using a 3/4 ton scroll compressor. Taking into account the heat exchange between refrigerant and oil, the performance characteristics have been numerically predicted.

A prototype of a small scroll compressor with high pressure side housing has been developed, utilizing the investigation results of oil influence on performance. Its characteristics have been examined by varying the operation speed over a range of 30 to 150*Hz*.

The total efficiency of a small scroll compressor both with tip seal and oil seal is comparable to that of a rotary compressor of equivalent capacity, and in particular the noise and vibration are lower than those of a rotary compressor.

INTRODUCTION

Recently in Japan, due to their high efficiency, low noise and minimal vibration, scroll compressors have come into use for medium size (2.5 to 5 tons) air conditioners, [1],[2]. On the other hand, for small size (below 1.5 tons) residential air conditioners, rotary compressors are widely used.

Generally speaking, due to the lower noise and vibration, the application range of scroll compressors can be extended to a smaller size, if the total efficiency can be increased by minimizing the leakage loss. Therefore, how to reduce the leakage loss has become the key to success of small scroll compressors. Some of the measures to reduce the leakage loss are oil injection, precision manufacturing and design of sealing mechanism such as a tip seal. For a small scroll compressor, however, we can hardly find studies on characteristics of those measures.

In this paper, the authors focus their attention on the sealing effect of injected oil which is very effective to minimize the leakage loss. The sealing effect of oil has been experimentally investigated, and the performance characteristics have been numerically predicted, taking into account the heat exchange between refrigerant and oil.

Using a prototype with high pressure side housing, characteristics such as performance, noise and vibration, are compared with those of a rotary compressor of equivalent capacity.

NOMENCLATURE

C_p = isobaric specific heat of refrigerant

C_o = specific heat of oil

G_r = mass flow rate of refrigerant circulating in system

G_o = mass flow rate of oil supplied into suction chamber

h = enthalpy of refrigerant in control volume

h_i, h_o = enthalpy of refrigerant entering, leaving control volume

m = mass of refrigerant in control volume

m_i, m_o = mass flow of refrigerant entering, leaving control volume

p = pressure in control volume

Q = heat transfer from refrigerant to oil in control volume

Q_{me} = thermal equivalent of friction loss

t = time

T_{g1} = temperature of refrigerant into suction chamber

T_{g2} = temperature of thermal equilibrium in suction chamber

T_{g3} = mean temperature of discharge gas

T_o = temperature of oil into suction chamber

u = internal energy of refrigerant in control volume

V = volume of compression chamber

v = specific volume of refrigerant in control volume

W_{ad} = adiabatic compression work

X = oil quantity ratio

η_i = indicated efficiency

INFLUENCE OF OIL QUANTITY RATIO ON PERFORMANCE

Sealing Effect Of Oil

The sufficient oil quantity necessary to seal has been investigated using the experimental apparatus shown in Fig.1. This apparatus has a 3/4 ton scroll compressor manufactured precisely, utilizing a tip seal.

The change of input power was obtained by varying the oil quantity ratio to suction gas. From the test results, we find that in order to reduce input power, it is necessary to maintain an oil quantity ratio above 8%, and if the oil quantity ratio is insufficient, it requires more power as shown in Fig.2, where the oil quantity ratio X is defined as follows :

$$X = \frac{G_o}{G_r + G_o} \tag{1}$$

The measured *P-V* diagram is shown in Fig.3. If the oil quantity ratio is insufficient, the pressures are higher and the indicated loss is larger. Therefore, in order to improve the indicated efficiency of small capacity scroll compressors, it is necessary to maintain a large oil quantity ratio of more than 8%.

Suction Gas Superheating With Oil

In order to maintain sufficient oil quantity in the suction gas flow, the oil must be separated from discharge gas flow and be recycled. The recycled hot oil superheats the suction gas and reduces the volumetric efficiency. On the other hand, during the compression process the oil cools the gas, and as a consequence, the compression work is reduced. Therefore, it is necessary to estimate the reduction of compression work and volumetric efficiency by the superheating loss. They are computed with a thermodynamic model taking into account the heat balance between refrigerant and oil. The scheme of this model is shown in Fig.4. The following is an outline of this model.

Some assumptions, as follows, are applied for simplification :

(1) Fluid properties are uniform throughout the volume.
(2) Kinematic energies are neglected.
(3) Instantaneous mass flow is considered as one-dimensional, steady flow in a converging nozzle.
(4) The heat transfer coefficient between gas and oil is so large that both come rapidly to thermal equilibrium.
(5) The oil quantity ratio is identical at any control volume.
(6) The volume occupied with oil is considered negligible.

The first law of thermodynamics for a control volume can be written as

$$d(m \cdot u) = -dQ - p \cdot dV + \sum (dm_i \cdot h_i) - \sum (dm_o \cdot h_o) \tag{2}$$

and the continuity equation is the following.

$$dm = \sum dm_i - \sum dm_o \tag{3}$$

Using these fundamental equations, we can get the following equation for the rate of change of enthalpy in the control volume :

$$\frac{dh}{dt} = \frac{\left\{-\frac{dQ}{dt} + \sum (h_i - h) \cdot \frac{dm_i}{dt} + v \cdot \left(\frac{dV}{dt} - v \cdot \frac{dm}{dt}\right) \cdot \left(\frac{\partial p}{\partial v}\right)_{h=\mathrm{constant}}\right\}}{m \cdot \left\{1 - v \cdot \left(\frac{\partial p}{\partial h}\right)_{v=\mathrm{constant}}\right\}} \tag{4}$$

This equation can be solved using real gas properties of R22.

The assumed temperature of the recycled hot oil can be obtained by adding the temperature rise which is caused by the thermal equivalent of friction loss to the mean discharge gas temperature T_{g3},

$$T_o = T_{g3} + \frac{Q_{me}}{G_o \cdot C_o} \tag{5}$$

The friction loss was obtained from the bearing system analysis of a typical scroll compressor. Coulombic friction was assumed to take place where boundary type lubrication occurs, such as in the thrust bearing. It was considered that hydrodynamic lubrication takes place in journal bearings. We referred to the literature [3] and [4] for a detail of the bearing system analysis.

The heat exchange between suction gas and recycled oil takes place in the suction chamber. The temperature of thermal equilibrium is obtained from the following equation.

$$G_r \cdot C_p \cdot (T_{g1} - T_{g2}) + G_o \cdot C_o \cdot (T_o - T_{g2}) = 0 \tag{6}$$

As its temperature rises, the suction gas is rarefied and the gas enters the compression chamber. While being cooled by oil, the gas is compressed, then it passes through the discharge port into the discharge chamber. The compressed gas leaves the compressor and the oil is separated from the gas in the discharge chamber and returns to the suction chamber again.

The results calculated with this analytical model are shown in Fig.5. The suction gas is superheated by recycled hot oil. The increment of oil quantity ratio causes the suction temperature to rise and, as a consequence, the volumetric efficiency decreases. On the other hand, due to the cooling effect of the oil, the compression work decreases slightly with the increment of oil quantity ratio. The total efficiency is approximately 4.5% lower at the oil quantity ratio of 10% than at 0%.

STRUCTURE

Hermetic scroll compressors with either a high or low pressure side housing are both in production today. In the case of high pressure side housing, as much oil as needed can be supplied because the oil is easily separated in the housing. Considering the sealing effect and the superheating loss, it is necessary for small scroll compressors to supply a proper amount of oil to the suction gas flow. Therefore, we have determined to develop a high pressure side housing for the prototype with 13.2*cc* displacement.

Scroll compressors have 2 kinds of leakage path. One is a radial leakage path which allows gas to flow through the tip clearance (T-GAP), and the other is a tangential leakage path which allows gas to flow through the flank clearance (F-GAP). It is desirable to minimize these clearances. In general, the influence of T-GAP leakage on performance is greater than that of F-GAP leakage [5]. Therefore, in this investigation, we have adopted a fixed crank method which depends on high precision machining and assembly. In order to minimize T-GAP leakage, a tip sealing method has been adopted.

Total efficiency was examined by varying the oil quantity ratio using this compressor. The results (Fig.7) show that a higher efficiency is obtained at an oil quantity ratio between 8 and 10%.

PERFORMANCE CHARACTERISTICS

The performance characteristics over a wide range of speed have been analyzed by using methods such as measuring the pressures of compression chambers. The results of efficiency evaluation are shown in Fig.8. Indicated efficiency ηi is defined as follows :

$$\eta i = \frac{Wad}{\oint p \cdot dV} \tag{7}$$

Mechanical loss is obtained by subtracting the flow, thermal, leakage and motor losses from the total loss. Motor efficiency was obtained from the test results of the motor itself.

It is summarized as follows :

(1) Over the speed range of 60 to 150*Hz*, the indicated efficiency is maintained at a high level, due to the sealing effect of both oil and tip seal.

(2) Within the range of low speed, the volumetric efficiency decreases, due to the increase of superheating loss affected by the recycled hot oil. This is because the oil quantity ratio increases as the speed decreases, since the recycled oil is maintained nearly constant regardless of the speed.

COMPARISON WITH ROTARY COMPRESSOR OF EQUIVALENT CAPACITY

Loss Comparison

The distribution of scroll compressor losses has been compared with that of rotary compressor losses of equivalent capacity. The results are shown in Fig.9. In brief,

(1) The indicated loss of a scroll compressor is less than that of a rotary compressor. This is because the over-compression loss of the scroll compressor is less as shown in Fig.10. This is due to a continuous flow during the discharge process.
(2) The mechanical loss of a scroll compressor is greater than that of a rotary compressor, mainly because the thrust bearing frictional loss is large.
(3) The total efficiency of a small capacity scroll compressor is comparable to that of a rotary compressor.

Noise And Vibration Comparison

The sound spectrum test results (Fig.11) show that the sound pressure level of a scroll compressor in the frequency range of 500 to 5000*Hz* is much lower than that of a rotary compressor. The overall sound pressure test results (Fig.12) show that the sound pressure level of a scroll compressor is 10*dB* lower over the speed range of 30 to 150*Hz*.

In addition, it was obtained experimentally that the vibration is lower than that of a rotary compressor, especially in a low speed range as shown in Fig.13. Due to the rotary compressor characteristics of compression torque fluctuation, the rotational velocity is affected and fluctuates. Therefore, its torsional vibration is much larger than that of a scroll compressor [6].

CONCLUSIONS

A scroll compressor with high pressure side housing has been developed for residential air conditioners and its characteristics have been investigated.

The test results show that the total efficiency of a small capacity scroll compressor with both tip seal and oil seal is comparable to a rotary compressor of equivalent capacity. The results also show that the noise and vibration characteristics of a small scroll compressor are, as predicted, superior to those of a rotary compressor.

The influence of oil on performance has been analyzed experimentally and numerically. For small capacity scroll compressors, the oil quantity ratio must be controlled to obtain sufficient oil seal and eliminate excessive superheating of suction gas.

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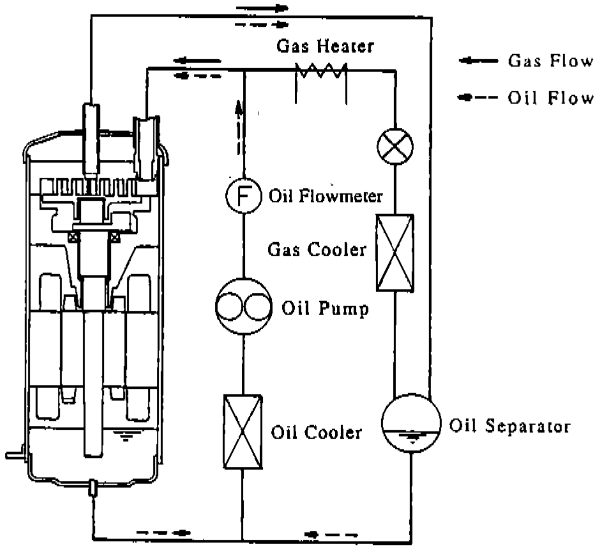

Fig. 1 Schematic Diagram of Oil Injection Test

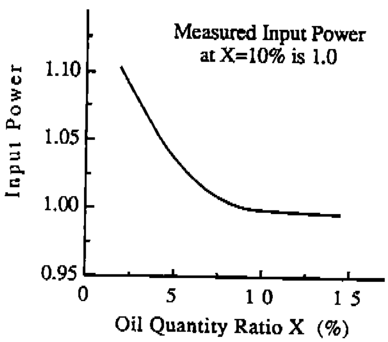

Fig. 2 Input Power vs.
Oil Quantity Ratio

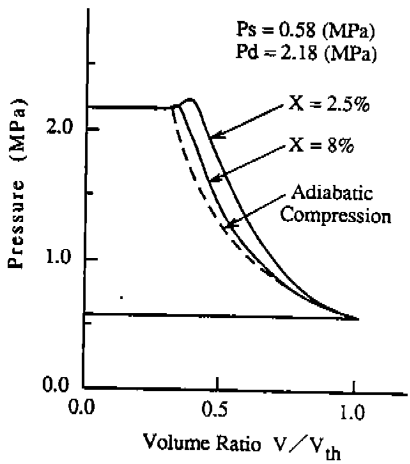

Fig. 3 P-V Diagram

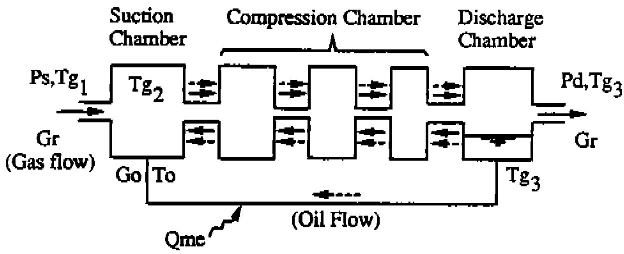

Fig. 4 Analytical Model

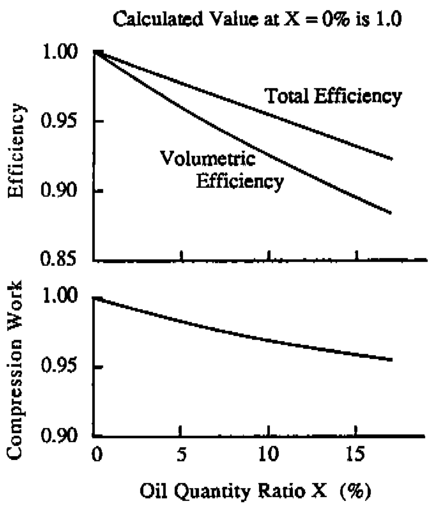

Fig. 5 Influence of Oil Quantity Ratio on Performance (Calculated)

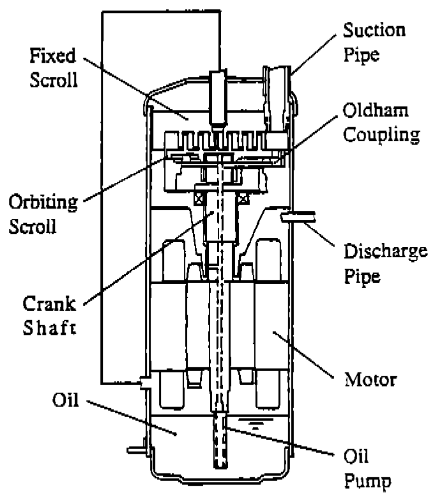

Fig. 6 Schematic View of Scroll Compressor

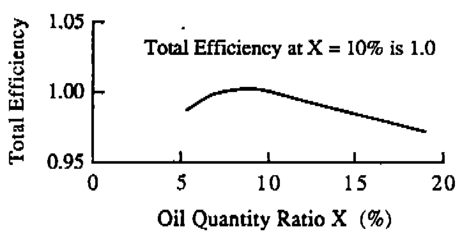

Fig. 7 Total Efficiency vs. Oil Quantity Ratio

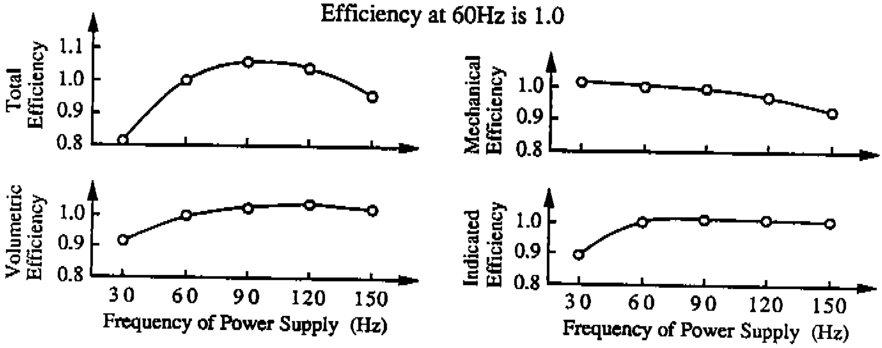

Fig. 8 Efficiency Evaluation

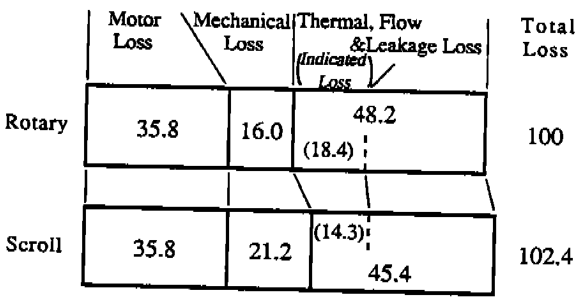

Fig. 9 Loss Comparison

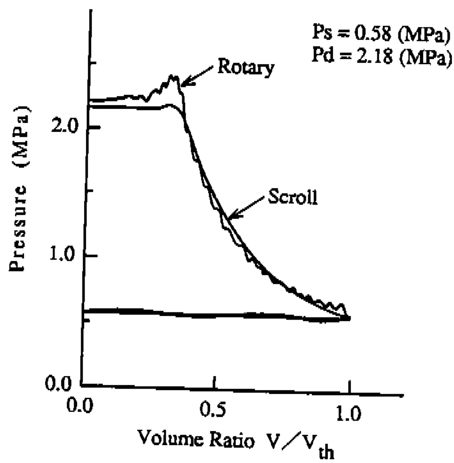

Fig. 10 P-V Diagram

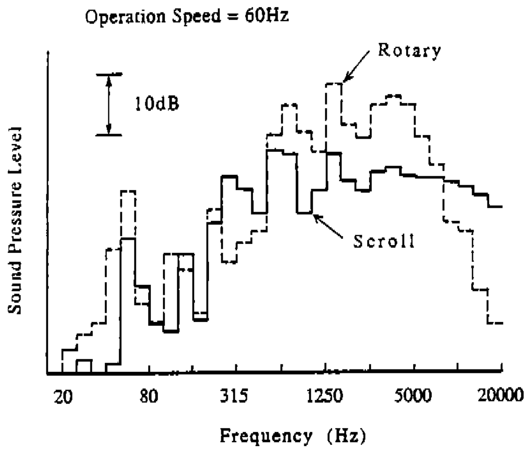

Fig. 11 Sound Spectrum in One-third Octave Band Analysis (A weighting)

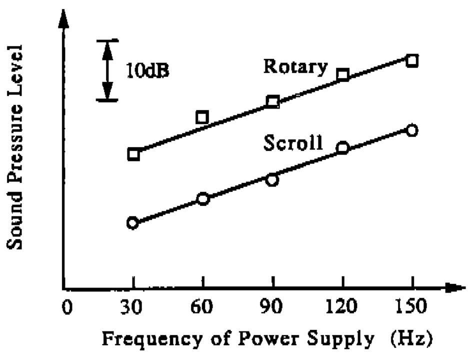

Fig. 12 Sound Pressure Characteristics (A weighting)

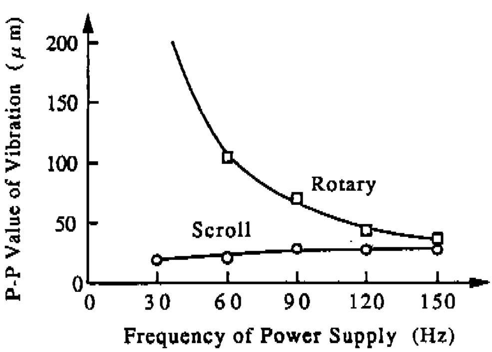

Fig. 13 Vibration Characteristics